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ABSTRACT

The main methods of screw compressor capacity control are shaft speed variation, suction throttling and variation of the suction volume. These have been analysed and are compared by the use of a standard mathematical model which has been adapted for this purpose. The analysis is based on a screw compressor with 5/6 and 5/7 "N" rotor profiles for air and refrigeration compressors. The results are compared with experimental data for variable speed of rotation and throttling operation obtained from an air compressor.

INTRODUCTION

Screw compressors are used for a variety of applications. For optimum performance from such machines, a specific design and operating mode is needed for each application. In the case of refrigeration plant, they are expected to operate efficiently over a wide range of loads for long periods. As was emphasized by *Koelet, 1992* and *Wang, 1993*, capacity control is one of the most important aspects of refrigeration compressor operation.

The analysis of the compressor behaviour under part load operation has been facilitated by recent advances in mathematical modelling and computer simulation. By this means, the earlier approach of process analysis and optimisation by intuitive changes, verified by tedious trial and error testing, can be eliminated. As a result, the design of screw compressor flow control systems has substantially evolved over the past few years and is likely to lead to further system performance improvements in the near future. The use of such methods is reported by *Pillis, 1986.*

A long term research programme in positive displacement compressor technology at the Mechanical Engineering Faculty of Sarajevo University, led to the development of a suite of subroutines for predicting the thermodynamic processes in screw compressors which have been verified by extensive comparison with experimental measurements. Details of this are given by *Hanjalić and Stošić, 1997.* Additional features have been incorporated into this analytical model to study such effects as variable built-in volume and economizer mode of operation on the capacity control on refrigeration systems. More information of the mathematical modelling of these effects can be found in *Jonson, 1988* and *Sjöholm, 1986.*

Two rotor configurations of the same main rotor size; namely a 5/6 and 5/7 of 127.5mm diameter, as shown in Fig. 1, were considered. The 5/6 arrangement was first analysed for the design of an air compressor which was then built and tested. Once its estimated performance was verified experimentally, both configurations were used for a study of refrigeration compressor capacity control systems. The rotors shown are typical members of a family of rack generated involute 'N' profiles designed for the efficient compression of air, common refrigerants and process gases for which patents *Stošić, 1996* have been granted. The profile is defined on the rotor rack as follows: $E-F$ is a circular arc on the rack, $F-G$ is a straight line for the upper involute. $G-H$ on the rack is a trochoid generated by a small portion of the arc F_2-H_2 on the gate rotor, while $H-A$ on the rack is another trochoid generated by the small portion of the arc A_1-H_1 on the main rotor. $A-B$ is a parabolic arc on the rack. This ensures both a large rotor displacement and a low torque on the gate rotor. $B-C$ is another straight line on the rack needed for the lower involute, while $C-D$ is a circular arc on the rack. Segment $D-E$ is a straight line for the rotor inner/outer circles. This forms a template for a variety of profiles which are a compromise between the needs for full tightness, small blow-hole area, large displacement, short sealing lines, small confined volumes, involute rotor contact and appropriate torque distribution, as well as high rotor mechanical rigidity. Further details of these rotors are given by *Stošić and Hanjalić, 1997*.

METHODS OF ANALYSIS APPLIED FOR CAPACITY CONTROL

Mathematical Modelling

A set of equations which describe the physics of the complete set of processes in a compressor was used here to calculate the effects of capacity control on compressor performance. It comprises the conservation equations for energy and mass continuity, as well as the momentum equation, supported by a number of differential or algebraic equations, to define leakage, fluid injection and other accompanying phenomena. By using local fluid properties, supplied from separate routines, for each position in the machine, given the rotation angle, and incorporating the local values of leakage flows, friction losses and oil-gas interaction, the modelled equations can be integrated to evaluate all thermodynamic and fluid properties as functions of the rotation angle. The resulting system of equations, applied over a compressor cycle with appropriate initial and boundary conditions, is solved by means of the fourth order Runge-Kutta numerical method to include fluid suction, compression and discharge. Due to the iterative nature of the numerical procedure, several, but usually less than four, iterations are required until the desired convergence is achieved. As a result, the trapped mass, pressure and temperature in the working chamber are calculated as functions of the rotation angle. These are then used in further calculations to obtain the $p-V$ diagram, compressor fluid and oil flow rates, turbulence and heat transfer. The results are then integrated over the cycle to estimate overall machine performance characteristics such as power input, specific power, volumetric and adiabatic efficiencies and hence the C.O.P..

Compressor Measurements

A compressor with a 5/6 rotor configuration, which was designed by the authors for an air compressor manufacturer, was used as a test vehicle for the modelling and measurement of the air compressor capacity control. The rotors were machined by a specialist rotor manufacturer while the other components were either machined by jig and tool manufacturers, who were not specialists in screw compressor technology, or, in the case of bearings and seals, purchased as standard parts. A full description of this design is given by *Stošić at al, 1997*.

A compressor test rig was designed and constructed at the City University Compressor Centre Laboratory and certified by Lloyd's Register as fully compliant with PNEUROP/CAGI 1992, PN2 CPT C1 test codes for bare displacement air compressors. This uses a 100 kW Diesel engine to meet both the requirements of variable speed operation and limitations on the laboratory electricity supply and permits the testing of liquid-flooded air compressors of up to 16 m^3/min. High accuracy instruments were used to make direct measurements of atmospheric, suction, discharge and flow orifice plate pressures, orifice plate pressure difference, suction, discharge, orifice plate and oil injection temperatures together with compressor rotational speed and torque. These were then recorded and processed in a data logger. By this means two sets of data were obtained. Measured values were used to calculate the air flow rate, power and specific power. The oil injection rate was estimated by means of a heat balance. The test results were compared with the mathematical model values and used to validate and tune the mathematical model.

DISCUSSION OF THE RESULTS

Air Compressor

Results for a typical capacity control of an oil flooded air compressor are presented. Two methods were investigated both analytically and experimentally, namely: speed variation and suction throttling. The effect of slide valve control, which is not typical for air compressors, was also calculated. A comparison between experimental and calculated results for both speed variation and suction throttling is given in the form of flow, power and specific power in Figs. 2 and 3 respectively. Fig. 4 shows the influence of inlet volume variation on screw compressor performance. As can be seen, speed variation and slide valve motion have

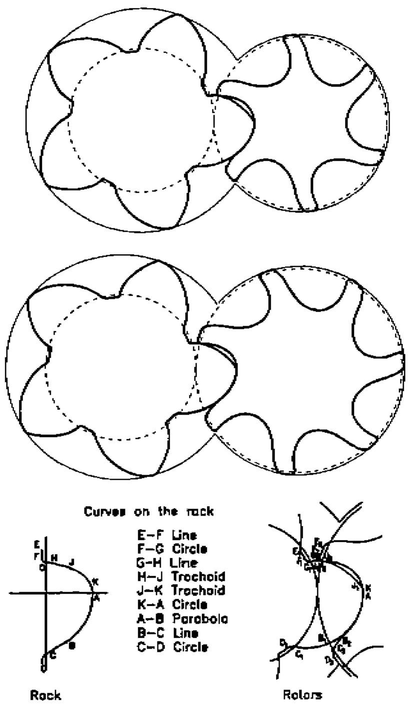

Fig. 1: 5/6 and 6/7-127.5 mm Rack Geneated 'N' Screw Compressor Rotors

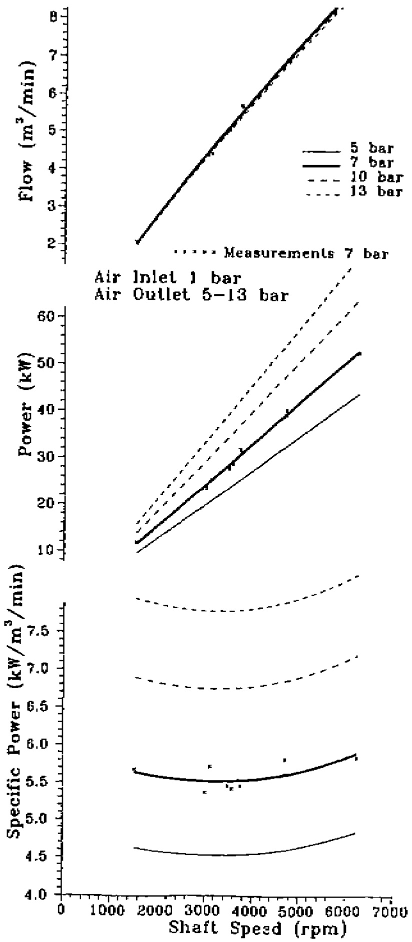

Fig. 2: Air Compressor Performances as a Function of Shaft Rotational Speed

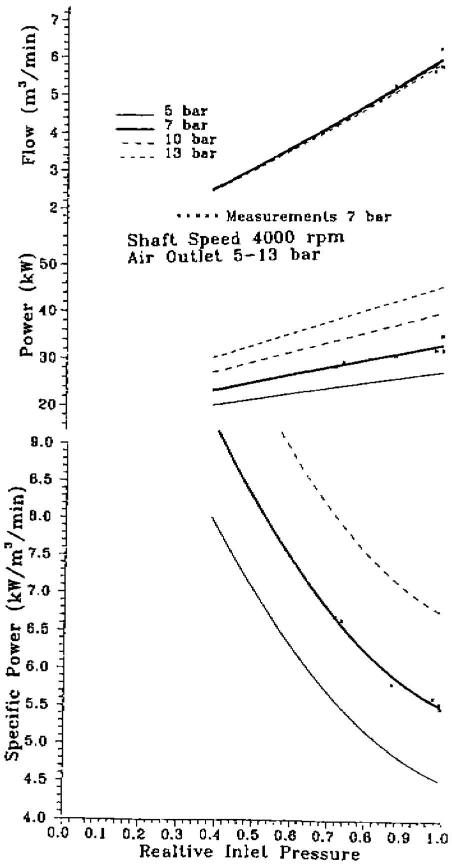

Fig. 3: Air Compressor Performances as a Function of Suction Throttling

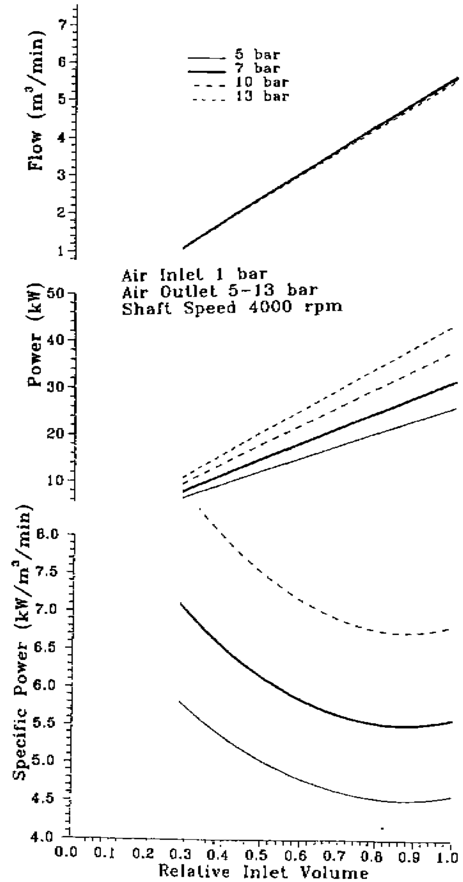

Fig. 4: Air Compressor Performances as a Function of Inlet Volume

a similar effect, while throttling degrades the compressor performance, especially when the flow is reduced significantly.

Refrigeration Compressor

A simple refrigeration plant was considered as a test case. The working fluid is $R134A$. When a refrigeration plant operates on full load, the condenser and evaporator heat exchange surfaces will be fully utilized. At part load, the heat transferred will be lower and hence the condensation temperature will decrease and the evaporation temperature will increase due to the excess area of the heat transfer surfaces. The compressor operating pressure ratio will therefore increase but since the changes in temperature are not large, the adverse effect of this on the compressor power will be small. Three different capacity control methods are analyzed by the mathematical modelling: speed control, throttling by means of a suction control valve and suction volume variation by means of a compressor slide valve.

A shaft speed of 4500 rpm and an evaporation temperature of -5^oC, with unthrottled suction and with the slide valve fully open were taken as the plant base design parameters. For capacity control purposes, the speed is varied from 1000-10000 rpm. The suction pressure throttling ratio is varied from 0, corresponding to total valve closure, to 1, when the valve is fully open. The slide valve is positioned between zero and full suction volume. Three additional evaporation temperatures were considered, namely: 5^oC, -20^oC and -35^oC. These cover the cases of home and commercial refrigeration and industrial low temperature refrigeration. 5^oC superheat for compressor suction and a constant condensation temperature of 35^oC were assumed for all cases. A few representative diagrams are presented in this paper while the complete set of results is given by *Kovačević 1997*.

In the case of variable speed capacity control, compressor mass flow and power are almost linear functions of speed. Their departure from linearity is due to variation in the adiabatic efficiency. Volumetric and adiabatic efficiencies only show significant change at speeds below 30% of the design value, when leakage losses start to become significant. Thus, variable speed, where achievable, will give the compressor excellent part load characteristics. Unfortunately, variable speed electric motors are still too expensive to be widely used in refrigeration plant. Nonetheless they are steadily becoming cheaper and Fig. 5 demonstrates refrigeration capacity and C.O.P as functions of shaft speed for four different evaporation temperatures.

Suction throttling is a convenient and effective means of capacity control. The throttle valve causes an additional pressure drop during compressor suction. The suction valve variable position is incorporated in the model by treating suction pressure drop as an input parameter. The suction pressure drop causes a steep decrease in the mass flow and usually a small decrease in the compressor power. The power is not proportional to the mass flow, because the suction pressure drop will also increase the compression pressure ratio which increases the power. Fig. 6 gives the refrigeration capacity and C.O.P as functions of suction throttling for three different compressor speeds. It should be noted that the power input at zero flow is about 60% of the full flow power. Suction throttling is effective, but not an economic means of capacity control. Continuous suction throttling can therefore be recommended only for short operating periods when flows are not less than 70% of the design value. A preferable throttling control is therefore to operate in the "on-off" mode so that the compressor rotates continuously but flow through it is intermittent; especially since this will not affect the oil flow through the compressor which is governed by the pressure difference across it.

A simple slide valve causes the compressor built-in volume ratio to change during its operation which introduces additional thermodynamic losses at all loads other than the design point. Fig. 7 shows refrigeration capacity and C.O.P. as functions of slide valve position for three different compressor speeds. An improvement on this is to use a "variable" built-in volume slide valve which keeps the built-in volume ratio more or less constant over the whole range of operation and hence minimises part load thermodynamic losses but this was not considered in the analysis.

A comparison of the effects of all three types of flow control on C.O.P. plotted for one evaporation temperature, -5^oC, is shown in Fig. 8. It follows from this and the other analyses considered, that the effect of capacity control on the screw compressor working process for a single stage refrigeration cycle may be summarized as: as:

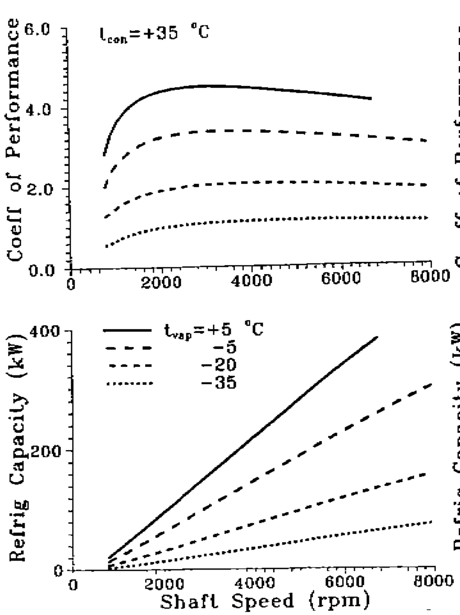

Fig. 5: Refrigeration Capacity and COP as a Function of Shaft Rotational Speed

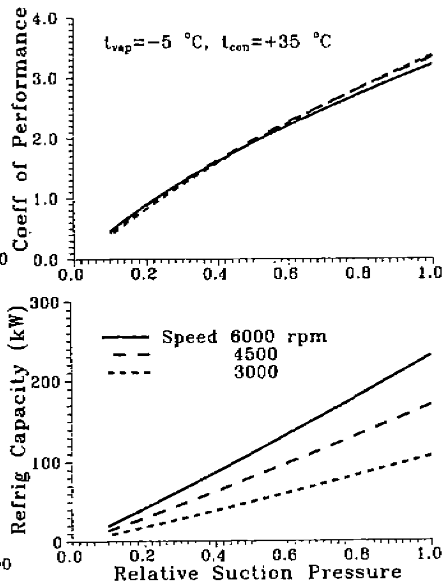

Fig. 6: Refrigeration Capacity and COP as a Function of Suction Throttling

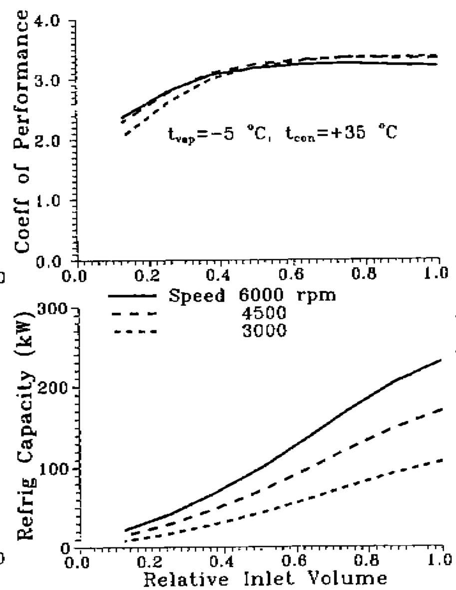

Fig. 7: Refrigeration Capacity and COP as a Function of Inlet Volume

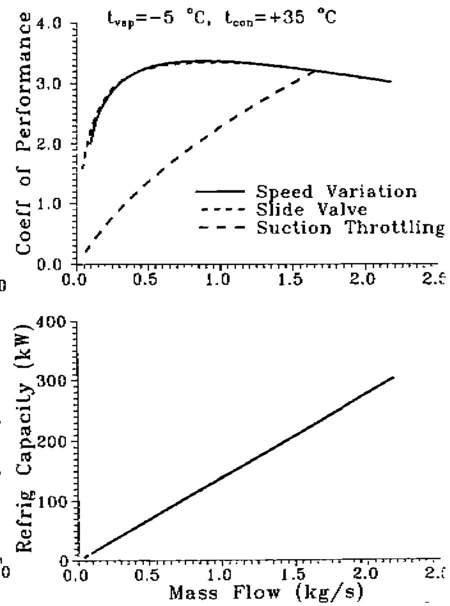

Fig. 8: Refrigeration Capacity and COP as a Function of Mass Flow Controlled by the Speed Variation, Suction Throttling, or Slide Valve Action

- For air conditioning or heat pump plants, when the evaporation temperature is between -5 and 5^oC, the most efficient method of load control is to vary the suction volume by a simple slide valve or its mechanical equivalent. In the case of high evaporation temperatures, this gives high refrigeration capacities for any mass flow, together with low specific power and high volumetric and adiabatic efficiencies.
- For industrial refrigeration plants with evaporation temperatures down to -20^oC, the effects obtained with a slide valve capacity control and a speed capacity control are similar. The choice depends only on the availability and cost of a variable speed drive.
- For low temperature refrigeration with evaporation temperatures of -35^oC, the best results are obtained by variable speed capacity control.
- Suction throttling capacity control can be used for each of these applications if the period of throttling is not excessive and if the load is not expected to change significantly.
- The use of a number of smaller compressor units instead of a single large unit is not analysed separately here, but if used properly, it can minimise the energy consumption because it results in the some of the compressor units operating at close to their design point. In this case, partial load is achieved by an on-off control for each compressor.

CONCLUSIONS

As screw compressors play an increasing role in refrigeration applications such as air conditioning, moderate and low temperature refrigeration and heat pumps, it is necessary to analyze their capacity control in order to use them most efficiently and cost effectively.

In general, it may be concluded that speed variation is the most efficient flow control method for all applications. Slide valve motion which results in suction volume variation is the next most efficient flow control method, especially for air conditioning and heat pump applications where the pressure ratios are not excessively high. Capacity control by suction throttling, is the simplest and cheapest method and can be used for short periods of throttling or for small load variations.

In the case of refrigeration plant fitted with an economiser, speed capacity control is still the best flow control method. In this case, a slide valve capacity control is only efficient for small load variations. This is highly dependent on the position of the economizer port. Suction throttling capacity control gives better results for economizer plant than it does for a single stage refrigerating compressor system without it but it is best applicable only for small capacity changes and short periods of throttling.

REFERENCES

Hanjalić K and Stošić N, 1997: Development and Optimization of Screw Machines with a Simulation Model, Part II: Thermodynamic Performance Simulation and Design Optimisation, ASME Transactions, Journal of Fluids Engineering, Vol 119, 664

Jonsson S, 1988: Performance Simulations of Twin-Screw Compressors with Economizer, Purdue University Compressor Technology Conference, West Lafayette, 884

Koelet P.C, 1992: Industrial Refrigeration, Design and Applications, Marcel Decker Inc, New York

Kovačević A, 1997: Analysis of the Capacity Control Upon the Refrigeration Compressor Process, Thesis for the degree of MSc, University of Tuzla, Bosnia and Herzegovina

Pillis J.W, 1986: Advancement in Refrigeration Screw Compressors Design, ASHRAE Transaction, Part IB, 219

O'Neill P.A, 1993: Industrial Compressors, Theory and Equipment, Butterworth-Heinemann, Oxford

Sjöholm L, 1986: Variable Volume-Ratio and Capacity Control in Twin Screw Compressors, Purdue University Compressor Technology Conference, West Lafayette, 494

Stošić N, 1996: UK Patent Application GB 9610289.2

Stošić N., Hanjalić K., 1997: Development and Optimization of Screw Machines with a Simulation Model, Part I: Profile Generation, ASME Transactions, Journal of Fluids Engineering, Vol 119, 659

Stošić N, Smith I K, Kovačević A, Aldis C, 1997: The Design of a Twin Screw Compressor Based on a New Rotor Profile, Int. Journal of Engineering Design, Vol 8, 389

Wang S.K, 1993: Handbook of Air Conditioning and Refrigeration, McGraw-Hill Inc, New York